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HIGH-SPEED OIL ENGINES FOR VEHICLES

By Ludwig Hausfelder

PART III

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PART III.*

As an example of an ignition-chamber engine, I will mention the Diesel engine of Benz & Company, which was first exhibited at the 1924 Berlin Automobile Exposition, and which was installed in a motor truck (Fig. 27). The engine, whose combustion chamber differs but little from that of the Mannheim stationary engine, had four separate vertical cylinders with a 125 mm (4.92 in.) bore and a 180 mm (7.09 in.) stroke; furnished about 45 HP. at $n = 1000$ R.P.M.; and was started by an auxiliary motor with ventilated valves. The first ignitions were produced by means of an electrically heated spiral wire situated in the ignition chamber. The truck, which was equipped with this engine, was not exhibited at the 1925 exposition, so that it may be assumed that the engine has not yet been satisfactorily developed as a vehicle engine. On the other hand, the two-cylinder engine made by the same company has proved very satisfactory, especially for motor plows, tractors, field-railway locomotives, etc. In spite of the fact that the attendance and

*"Schnellaufende Oelmotoren für Kraftfahrzeuge." From "Der Motorwagen," December 20 and 31, 1926, and January 10, 1927. For Parts I and II, see Technical Memorandums Nos. 397 and 403.

care cannot be very good with such uses, it has given no cause for complaint. To be sure, it can be termed neither a high-speed nor a very light engine - with a bore of 135 mm (5.31 in.) and a stroke of 200 mm (7.87 in.) it furnishes 30 HP. at n = 800 and weighs about 30 kg (66 lb.)/HP., - but it exhibits such a similarity to the modern vehicle engine, that it should be mentioned at this point. The upper part of the crank case is cast from aluminum in one piece with the cylinders and the cylinder linings are exchangeable cast-iron bushings. The lower part of the crank case and the separate cylinder heads are cast iron. Further details are shown in Figs. 28-29, which (for the lack of German sources) had to be taken from "Automotive Industries," of October 29, 1925. The final compression pressure is 32 atmospheres and the pump pressure about 70 atmospheres. It is started by hand, the ignition being produced by a nitrated paper cartridge which is introduced through a hole into the ignition chamber, the cartridge being held securely in place by a bayonet fastening.

A modern vehicle engine, working on the mechanical-injection principle, is the light M.A.N. Diesel engine, which develops about 50 HP. at n = 1000 R.P.M., with 115 mm (4.53 in.) bore and 180 mm (7.09 in.) stroke. Its weight, including the flywheel, is about 500 kg (1102 lb.) or about 10 kg (22 lb.)/HP. The cylinders are cast in a block, rest on an aluminum crank case, and have detachable heads for each pair of cylinders. The hanging

valves are actuated in the usual way, through push rods and rocking levers, by two cam shafts in the crank case (Figs. 30-31). For each cylinder there are two single-hole injection nozzles which are supplied by the same pump (Fig. 32). The engine can be started from the cold condition, either by hand crank or by the electric starter without any special auxiliary fuel or other means for producing ignition. The lubrication and water circulation correspond throughout to the systems commonly employed in carburetor engines. Outwardly the Diesel engine differs from the carburetor engine only in the substitution of the fuel pump and nozzles, with the necessary piping, in place of the carburetor, spark plugs, and magnetos.

These parts are entirely new to the automobile constructor and are all the more difficult to make, because there is no description or pattern to start from. Although it was not possible to apply the knowledge obtained from experience with stationary engines to high-speed vehicle engines without critical investigation, it was still less practicable to adapt these parts, derived from large Diesel engines, to small vehicle engines simply by proportionately reducing their dimensions. On the contrary, comprehensive theoretical and experimental research was necessary in order to discover suitable foundations for the designing of these parts and to construct practically utilizable types. It would be erroneous to claim that these researches are finished. On the contrary, we are still in the midst of them

and far from their completion. The following description of nozzles and fuel pumps can therefore give only a very imperfect picture of the present stage of development.

Fuel-Injection Nozzles

As already stated in the description of the injection process, the atomization depends chiefly on the atomization pressure and the kind of nozzle used. The chief requirement of a nozzle is to produce as unbroken a jet as possible, coupled with maximum penetrative power and fineness of atomization. Since the penetrative power of the jet is increased by the use of long tapering channels, while on the contrary, the fineness of the spray is increased by short channels and suitable shapes of the nozzle outlets, we must generally accept compromise solutions. From the numerous kinds of nozzles, whose abundance is indicated by the greatly increased number of patents in this field, it is obvious that constructors are trying to find the best combination of these conflicting conditions.

Nozzles can first be classified as open and closed, according to whether the fuel pipe is shut off from the combustion chamber (except during the actual injection period). They can then be classified, according to their cross sections, as round-hole nozzles (single-hole and multi-hole nozzles), slot nozzles and annular nozzles. When it is considered that most of the nozzle forms mentioned can be either open or closed nozzles and that,

in many of these nozzles, there is the further possibility of imparting a rotary motion to the fuel jet by means of spiral grooves (whirling-spray nozzle), the difficulty of a systematic and critical valuation of all the nozzle forms can be comprehended. I must therefore restrict myself to the description of a few typical nozzles, which are either of special importance for vehicle engines or specially characteristic of the development of the compressorless or "airless" Diesel engine. Of the closed nozzles, I will first mention the single-hole and multi-hole nozzles (Figs. 33-34) controlled by a conical needle. Due to the slight edge effect, the atomization is not very good, so that they will hardly find use in high-speed engines. Fig. 35 shows the original McKechnie nozzle. The fuel, which is divided by the longitudinal grooves in the valve spindle, is largely atomized on the sharp edges of the nozzle mouth and valve head as it leaves the nozzle. In the Peugeot-Tartrais nozzle (Fig. 36) the valve head is flat, so that the jet is affected only by the sharp edge of the nozzle mouth. On the other hand, Tartrais utilizes the whirling effect of the spiral grooves in the valve spindle for imparting additional velocity to the fuel jet through centrifugal acceleration. The increase in velocity thus obtained may be quite large. Comparisons by Woltjen (See N.A.C.A. Technical Memorandum No. 403, p.19 ff and Plate I, Figure f) show that the same degree of atomization can be obtained with a spirally grooved valve spindle (whirling-spray nozzle) at a pump

pressure of 150 atm., as with a smooth valve spindle at a pressure of 250 atm. This result is very valuable, because the reduction in the requisite pump pressure means not only a somewhat smaller consumption of energy in the operation of the pump, but also a much greater facility in keeping it tight. The fact, however, that the whirling-spray nozzle is now being less used is because it produces a very broad spray cone with less penetrating power than the narrower spray cones produced by other nozzles. On the other hand, the whirling-spray nozzle can be advantageously used, when the angle of the spray cone is varied according to the engine load. Fig. 37 shows the nozzle of the Lanz hot-bulb engine "Bulldog," in which the spirally-grooved valve spindle can be moved with reference to the nozzle mouth. When it is at a distance from the nozzle mouth, the effect of the spiral grooves is almost entirely eliminated and the nozzle produces a pointed spray, which is best suited to the conditions prevailing in the hot-bulb engine at a smaller load and lower revolution speed. The annular nozzle (Figs. 38-39) was probably designed originally to eliminate any possible fouling of the nozzle by the valve spindle. Another advantage is the guiding of the fuel for a longer distance and the greater penetrative power imparted to the fuel drops in a small jet.

Most modern nozzles are no longer operated mechanically by a cam shaft, but automatically through the fuel pressure itself. This pressure lifts the valve spindle against the closing force

of an externally applied spiral spring and leaves the nozzle cross section free. The fuel valve closes as soon as the fuel pressure falls below a certain point. The injection process can be so controlled by the suitable adjustment of the tension of the spring, that the fuel will be injected only after a certain pressure is reached. On again falling below a certain pressure, the injection will be shut off with corresponding quickness. It is thus possible to bring the injection process into a certain independence of the delivery curve of the fuel pump and largely to control, through a suitable combination of the valve-lift and pump-pressure curves, the injection process which determines the further course of the combustion line. If the control of the injection process is not effected by an adjustable spring, but is left to a special form of pump, a check valve, held by a weak spring, then suffices to protect the fuel-delivery pipe from the strongly fluctuating pressure conditions in the cylinder. An especially simple type of nozzle is obtained by constructing it so that it is automatically opened or closed by elastic deformation at certain pressure changes. Fig. 40 shows an older nozzle of the South Germany Engine Company at Küssnacht ("Acro") with an annular cross section. It consists of a stationary cylindrical or conical spindle firmly fitted into a round sharp-edged opening. The wide mouth produces a fine atomization, while the spindle guides the fuel jet. This nozzle therefore represents a very practical compromise. So far as any

information has been published, it has been successfully used in high-speed engines. Another nozzle which opens through elastic deformation is the "Lip" nozzle made by the Hannover "Waggonfabrik" (Figs. 41-42) in which two symmetrical or unsymmetrical flexible lips open so as to form a narrow slot and allow the fuel to escape in the shape of a fan. Here we also find extended outlet edges, though without any special jet guidance.

In all mechanically operated nozzles, which are closed by disk valves, spring valve spindles or elastic deformation, the outlet cross sections vary as the injection pressure but, when check valves are used, the cross sections remain constant during the injection period.

Nozzles which are shut off from the fuel pipes simply by check valves form the transition stage to the open nozzles, in which there is no valve between the outlet valve of the pump and the mouth of the nozzle. The fact that some of the leading firms are giving increased attention to the open nozzle is doubtless due to the advantages of this type, which make it very suitable for certain purposes and especially for small engines. The principal danger is that the open nozzles may become clogged from the combustion chamber side by oil and carbon deposits, and that their small outlets are more liable to obstruction by impurities in the fuel, than are the closed nozzles in which the deposition of foreign substances is generally prevented by the valve spindles. Although the impurities are removed as much as

possible by filtering, there is never any absolute guaranty against the clogging of the nozzles. The nozzles should be easily exchangeable and of simple design, so that they can be easily and quickly cleaned. The encrusting of the nozzles from the combustion chamber has not caused so much trouble as was anticipated. It is probable that the effective cooling of the nozzle by the fuel column prevents the adhesion of deposits. On the other hand, the open nozzle has the advantages that it can be easily and cheaply made, that the elimination of the valves removes a source of trouble, and that the air liberated from the fuel is automatically carried off under all circumstances.

The simplest possible nozzle is the single-hole nozzle (Fig. 43), which is used mostly in stationary engines. The impossibility, for structural reasons, of going below a certain bore often renders it impossible in high-speed engines to atomize finely enough the quantity of fuel injected. The same is true in a greater degree for multi-hole nozzles, which can only serve as central nozzles for large cylinders. Its use in vehicle engines is also opposed to Hesselman's statement that the ignition delay can be reduced to a minimum with a smaller number of holes (i.e., by stronger concentrations in the chemical sense). A better atomization is possible with slot nozzles than with round-hole nozzles, on account of the extended outlet edges of the former, which more thoroughly disperse the emerging jet. Fig. 44 shows an experimental nozzle, made by the South Germany

Engine Company, which consists of a tube pressed together so as to form a narrow slot. The fan-shaped jet is finely atomized by the rectangular edge. Another and more promising way to obtain very small drops consists in dividing the fuel inside the nozzle into two or more jets and in making these jets impinge on one another at a predetermined angle at the mouth of the nozzle. Fig. 45 shows a nozzle designed by Brenkert, in which the component jets strike one another at an obtuse angle. Even at moderate pressures the atomization is very complete and is much better than in single-hole nozzles at the same injection pressure. On the other hand, the directional and penetrative force of the jet is less since much the larger portion of the energy of the jet is expended in the work of atomization. This disadvantage is skilfully avoided in a nozzle made by Junkers (Fig. 46), in which the two component jets strike each other at an angle of 90°. The resultant of the velocities of the component jets is then in the direction of the emergent jet and imparts to the latter a sufficient directional and penetrative force. In designing this nozzle, special attention was given to facility of cleaning, by making the fuel channels in the form of mutually crossing grooves in the conical end of a stationary removable spindle.

The question, as to whether open or closed nozzles are preferable for heavy-oil vehicle engines, is difficult to answer and can be decided only in connection with the structure of the

engine in other respects, since the choice of the nozzle is closely connected with the shape and size of the combustion chamber, and the most favorable conditions can be determined only on the test stand. The greatest advantage of the open nozzle is its extremely simple and stable structure, which enables rapid replacement and cleaning. Furthermore, the previously mentioned automatic deaeration of the fuel pipe is of special advantage for vehicle engines, all the more because the shaking of the vehicles liberates more air bubbles than in stationary engines. On the other hand, there are other objections to their general adoption, aside from the possibility of the clogging of the open nozzle. For open nozzles, we have first to bear in mind that oil is considerably more compressible than water, and that the compressibility of the fuel, as well as the elastic expansion of the pipe, produces a measurable shifting of the injection instant. This delay in the beginning of the injection can be entirely eliminated by the proper adjustment of the pump. It is not so easy, however, to terminate the injection so as to prevent subsequent dripping from the nozzle, due to said elastic expansion. More will be said on this subject under the head of fuel pumps.

When it was stated in the description of closed nozzles that, especially in nozzles whose mouths were closed by heavily loaded disk valves or valve spindles, or by flexible "lips," the outlet cross sections vary in proportion to the injection pres-

sure, the characteristic was named, which forms the fundamental distinction between open and closed nozzles and can determine the choice of one or the other type. The constant outlet cross section of the open nozzle, for all pressures and outflow quantities, is often insufficient, at low revolution speeds and engine loading, to produce satisfactory atomization. On the other hand, the outlet cross sections of closed nozzles, corresponding to the lower injection pressure (due to greater leakage of the fuel pump) at low revolution speeds, open only just far enough to produce a good atomization through the increased friction of the closer edges of the nozzle mouth. The important characteristic of a good idling speed can therefore be obtained much easier with closed nozzles than with open nozzles.

Notwithstanding the fact that the vehicle Diesel engine of the M.A.N., which company has probably had the most experience with such engines, functions satisfactorily at idling speed with open nozzles, I still feel constrained to express the opinion that closed nozzles are, in general, the more suitable for vehicle engines. If they are made with flexible "lips," they are almost as simple as open nozzles. In this type there is, of course, the danger that, due to the cumulative effect of high combustion temperatures, the nozzle lips may partially lose their elasticity or be permanently deformed. I believe, however, that this defect can be reduced to a practically harmless magnitude by the use of suitable metal.

Fuel Pumps.

Perhaps the fuel pump makes the constructors of high-speed Diesel engines even more trouble than the correct choice and dimensioning of the nozzle. The pump has the important task of controlling the combustion process by regulating the time and course of the injection. If, in the dimensioning of the nozzle, theoretical problems are principally to be solved in close connection with thermodynamic problems, the designing of the pump then offers particularly important problems for the physicist. The construction of the pump makes great demands on the workshop and constitutes a criterion of the accuracy of the workmanship.

The difficulty of the problem is evident, when it is realized that, for a four-cylinder 50 HP. motor-truck engine, the pump has to deliver quantities of fuel varying from 30 to 170 mm^3 (0,00183-0.01037 cu.in.) at each piston stroke. These small quantities must be accurately measured and be injected at pressures of about 300 atm. without any considerable leakage, even at the minimum pump speed of $n < 200$ R.P.M. Furthermore, the pump must be of very simple and stable structure and protected against any possible admission and accumulation of air. Any excessive pressure, due to obstructions in the pipe or nozzle, must be eliminated.

Investigations need to be made as to which of the kinds of pump regulation employed on stationary compressorless Diesel

engines can be used on high-speed vehicle engines. The pumps in most common use are the ones with variable stroke, with overflow through a throttle opening, and with mechanically controlled discharge valves. The pump with overflow through a throttle opening (Fig. 47) is very simple and inexpensive. It is regulated, during the whole discharge stroke, by the return flow of the fuel, into the suction pipe through an opening whose free cross section is regulated by a needle valve. This pump, which has worked well on stationary engines, especially on account of its small back pressure on the regulator, can hardly be used on vehicle engines because, with the small quantity of fuel delivered, (i.e., with wide-open throttle cross section due to the diminished fuel velocity), it allows excessive quantities to pass over; in other words, because it is excessively regulated. Much better is the pump with variable discharge stroke, which is usually made with oblique-cam regulation (Fig. 48). The rapid wearing of the cylindrical or barrel-shaped tappet-guide rollers, which roll in point-contact on the oblique cams, is unsatisfactory. The pump with cam-guided intake or overflow valve (Fig. 49) must be regarded as the best solution. In this pump the effective stroke is suddenly interrupted by the opening of the suction or overflow valve at an instant determined by the governor and the discharge space of the pump is short circuited with the suction pipe.

The variable-stroke pump is suitable only for closed noz-

zles, in which a strong valve spring determines the beginning and end of the discharge stroke and therefore the portions of the pump-pressure curve in which the excess pressure of the fuel is increasing from zero and then falling back to zero. Since the nozzle bore is generally so calculated that it produces the best atomization at the maximum piston speed of the fuel pump, the atomization is often insufficient, especially at the beginning of the fuel delivery. Care must therefore be taken to obtain a rapid pressure increase. In so far as this is not possible through the suitable shaping of the cam, it can be helped by causing the cam to act, not directly on the pump piston, but on a push rod which, during its upward motion, pushes against the lower end of the piston and thus indirectly communicates the maximum velocity. This method has the disadvantage that the hard shock and the sudden pressure increase must be absorbed by the elastic yielding of the fuel pipe and the compression of the fuel itself. The rapidly oscillating waves of condensation and rarefaction thus produced in the fuel are reflected back and forth between the pump and nozzle and affect the injection process in a manner hardly to be anticipated. These pressure fluctuations are especially harmful toward the end of the injection period. If the needle valve does not close quickly enough or tight enough, there will be an after-dripping from the nozzle which, through after-burning, has a harmful effect on the combustion process and may cause incrus-

tation of the nozzle tip.

The pump with overflow regulation, in which the discharge stroke is suddenly interrupted by the opening of a valve, is therefore much better. The pressure equalization can be produced by opening the suction valve or an overflow valve in the compression chamber, which valve is connected with the intake pipe and is lifted by a tappet after covering the effective piston stroke corresponding to the position of the governor. If, as is usually the case with open nozzles, a very sudden pressure drop is desired, the cross sections opened by the intake valve will generally be too small for the fuel to expand quickly without excessive throttle losses. In this case, special overflow valves with rather large cross sections are almost exclusively used. A sudden removal of the pressure from the delivery pipe is facilitated by the fact that the quantity of oil subjected to the pressure change is made as small as possible by using short and small fuel pipes and also by the fact that the back-flowing fuel is not led directly into the suction pipe but into equalizing chambers which can receive large quantities of liquid quickly and without much resistance.

The disadvantages of suction and discharge valves which are not mechanically operated are not very noticeable at the hitherto customary revolution speeds of n = about 600 R.P.M. of fuel pumps designed principally for low-speed motor-truck engines. Difficulties in this connection need to be anticipated only when

the engine speed is increased. Even now, however, it seems desirable to operate the suction valve mechanically or to replace it by some other closing device. The reason for this lies in the fact that, on the one hand, the relatively weak spring of this valve does not guarantee a precise termination of the suction stroke and that, on the other hand, the reliability of the pump is endangered by solid particles getting between the valve and valve seat, which can hardly be avoided, even with the most careful filtering of the fuel. It has therefore been proposed to replace the suction valve by ports operated by the pump plunger. This method, which is said to have worked satisfactorily in stationary engines, might not, however, be so good for vehicle engines. The fact that the piston works in a vacuum during the first part of the suction stroke, favors the gasification of the fuel residue from the pump, as likewise the admission of air. The collecting of air or gas in the fuel pipe must therefore be absolutely prevented, since the compressibility of the gas in the fuel undoubtedly has an unfavorable effect on the injection process. The best solution would be mechanical operation through a piston valve, which should be so constructed that it would control both the suction and the discharge. The reliable tightening of the piston valve might be difficult, but not impossible.

The endeavor to produce a pump with mechanically operated closing valves led to the proposal (by Löwenthal and Egersdörfer

Conrad, and others) to use, as a fuel pump, the well-known rotary piston pump, which has long been successfully used as a spinning pump, especially in the manufacture of artificial silk (Fig. 50). The principle of this pump is based on the fact that two or more pistons, connected mechanically by a joint, rest on an oblique adjustable disk and, together with their cylinders, rotate about an axis outside the cylinder axis but parallel to it. A to-and-fro motion of the piston is produced by the following and sliding on the oblique disk. The piston stroke is altered by changing the obliqueness of the disk. The operation is changed by alternate connection of the rotating cylinder with the suction or discharge channels in a stationary cam plate. The rotary pump is classified, as a fuel pump, in the group of pumps with variable piston stroke and an approximately sinus-shaped delivery-pressure curve. Since, however, the latter is not desired, the pressure relations can be so controlled by suitable channel formation, as to produce a quick rise and fall of the delivery pressure. When it is further considered that the operation is entirely mechanical and that the rotary pump functions without back-pressure springs and that all the movable parts are sheltered by a simple cylindrical housing, it can be comprehended that the proposal of its use for the purposes of the high-speed Diesel engine met with hearty approval. Unfortunately, however, the rotary pump also has serious disadvantages. In the first place, the cylinder walls and pistons are strongly

stressed by the horizontal components of the piston pressure. Then the cylinder drum must be pressed firmly against the cam disk during the discharge stroke, in order to avoid leaks during the passage of the oil from the pump cylinder to the delivery channel. The unfavorable pressure curve, increasing in the form of a like-sided hyperbole asymptotically to the axis of rotation in the cam disk or in the frontal surface of the rotating drum, must therefore work itself out in unsymmetrical wear, which excludes any permanent tightness of the pump. Whether and to what extent these defects can be remedied by improvements such as those proposed by E. Müller (compression pistons between the drum and cam disk, intermediate levers for the prevention of horizontal piston pressures, etc.), must be determined by lengthy researches yet to be instituted.

All the above-mentioned pumps function with constant timing. For motor vehicles, however, there is often occasion for a change in the timing of the injection, in order to enable a smooth running of the engine, especially at a low revolution speed and at idling speed. It is true that the faulty atomization at a small load and low piston speed of the fuel pump automatically causes a certain ignition delay, but nevertheless the desired object can be much more accurately accomplished by changing the timing of the injection. In a normal pump, the beginning of the fuel delivery can be regulated within narrow limits by an adjustable intermediate lever tangential to the driving cam (Frey and

Fischer). A greater range of adjustability requires, between the pump and the driving shaft, a coupling which allows a relative rotation of the pump shaft.

At the present stage of development, no special rules can be formulated for the structural design of the pump. I must therefore confine myself here to a few general principles. Since the pump piston is almost exclusively moved forward mechanically by cams in the discharge stroke, but is nevertheless moved backward by a spiral spring in the suction stroke, it is impracticable to use stuffing boxes, whose friction might stop the piston. It is therefore important to bring both the cylinder and the plunger very carefully. At high liquid pressures, the accurate fitting of these parts and of the valves is absolutely necessary for the success of the engine. Nevertheless, especially at low revolution speeds, some leaks cannot be avoided, the effect of which must be offset by a corresponding excess of fuel. In order to obtain uniform wear, the occurrence of horizontal pressure components in the plunger must be avoided. The use of long push-rod guides or the introduction of rocking levers between the plungers and cams are suitable means for this purpose. Strong cylinder walls, which eliminate elastic deformation, and the smallest possible clearances naturally follow from the above considerations. In designing the pressure chambers and the arrangement of the valves, care must be taken to avoid the undesirable accumulations of air and gas, or at least

to conduct them harmlessly away, by providing suitable air valves in the pumps or discharge pipes.

For vehicle engines, which are predominantly multicylinder four-stroke engines, the individual pumps are grouped in a closed aggregate with a common intake pipe. They are generally driven by a short stub shaft with the cam shaft revolution speed. For pumps whose forward stroke or whose valves are actuated by oblique cams, it is expedient, according to Rumpler's proposal, to place the cylinders radially around the control cams (Fig. 51). The axial displacement of the cam simultaneously affects all pistons and valves in the rotation plane of the cam.

The structural difficulties naturally raise the question of the desirability of developing standard pumps and perhaps nozzles, as was long since done with magnetos and spark plugs. The fact that the fuel pump requires just as accurate workmanship as the magneto and that, moreover, certain parts of its driving gear (interrupter, automatic timing) can be advantageously used for the pump, indicates that the manufacturers of electrical ignition devices requiring the highest degree of precision are especially called to produce a standard pump. The interest taken by the Bosch Magneto Company (which has become the owner of the Acro patents by acquiring the stock of the South Germany Engine Company) in the development of the Diesel engine for vehicles, indicates that these views are entirely feasible.

There remains to be investigated as to how successful the

Diesel engine has been in comparison with the carburetor engine and as to what conclusions can be drawn regarding its use on vehicles. The general economical advantages of using native fuels has already been mentioned in the introduction. The advantages of their use for driving vehicles is best indicated by the following Table V, which gives the operation costs of a five-ton motor truck.

Table V

General expenses	5.6%	Fuel	14.0%
Operating "	14.1%	Oil	1.7%
Driver's salary	27.7%	Repairs	10.9%
Insurance & Taxes	4.0%	Depreciation	15.2%
Tires	6.8%		
		Total	100.0%

Since gas oil can now be bought for only one-fourth to one-third the price of gasoline or benzol, the fuel cost would be reduced to between 3.5 and 5% of the total expense. In arriving at this conclusion, the unfavorable assumption is made for the Diesel engine that its fuel consumption would be approximately equal to that of a carburetor engine. The calculated saving of 9-10% in the quantity of fuel consumed does not, however, cover all the advantages of using heavy oils. There is also a reduction in the fire hazard which is hard to express numerically. The difficultly ignitible heavy oils can be kept in plain casks without the necessity of purchasing expensive fireproof containers. The reduction of the fire hazard, both in the garage

and in use, will doubtless result in a reduction of the insurance premiums. When the vehicles are used intensively, as on motor-bus lines, the saving in the cost of operation is greatly in favor of heavy-oil engines. Here the proportionate expenses for containers, repairs, etc., diminish, while the expense for fuel may go as high as 30%. The use of heavy oils would reduce operation expenses about 20%, thereby greatly increasing the profits of such enterprises.

As already mentioned, the assumption that the fuel consumption of a Diesel engine is equal to that of a carburetor engine, is not quite accurate. In fact, the Diesel engine consumes only 190-220 g (0.419-0.485 lb.) per HP./hr., a value which is hardly attained by the best carburetor engines on the test stand, not to mention actual operation. Figs. 52-54 give the fuel consumption for several engines. The consumption of 185 g (0.408 lb.) per HP./hr., measured on the Acro engine at $n = 1380$ and 7-8 HP., which corresponds to a thermal efficiency of 34% ("Autotechnik," 1924, No. 26, p.22) is quite remarkable. It is an important fact for practical operation that the fuel consumption (per horsepower) remains nearly constant within a wide range of revolution speeds (Fig. 55), though it increases quite rapidly on carburetor engines with diminishing revolution speed. Both Diesel and carburetor engines consume about the same amount of lubricating oil. Due to more perfect combustion, the lubricating oil in the crank case is not thinned so much by the fuel in

a Diesel engine as in a carburetor engine. As regards care and attendance, no definite conclusions can be drawn from the comparison of the two engine types, due to the lack of sufficient experience in the operation of Diesel engines. In my opinion, however, the operation of a Diesel engine involves no especial difficulties and is not dependent on any special skill or technical knowledge of the driver. On the other hand, the attendance and upkeep of such an engine will require a special trained personnel until its peculiarities, as compared with the carburetor engine, are more generally known.

Aside from tractors, motor plows and locomotives, on which the Diesel engine has been successfully used for some time, its general use on motor trucks may be expected in the near future. By no means, however, will its use continue to be restricted to heavy vehicles, but it may be expected to enter soon into serious competition with the carburetor engine, even on light vehicles. The Dorner Oil-Engine Company of Hannover, has already made a light passenger car to carry about 250 kg (550 lb.) and equipped it with a compressorless Diesel engine (Fig. 56). The four-stroke-cycle mechanical-injection engine has two V-placed, air-cooled cylinders of 100 mm (3.94 in.) stroke and 70 mm (2.76 in.) bore and gives about 4.5 HP. at $n = 1400$ R.P.M. According to reports published in the technical press ("Wirtschaftsmotor" 1924, No. 11, p.6), the following data were obtained with gas oil having a specific gravity of $\gamma = 0.864$.

	Experiment 1 (normal load)	Experiment 2 (overload)
Revolution speed	1400 R.P.M.	1400 R.P.M.
Brake weight	3.800 kg .38 lb.	5.000 kg 11.02
Fuel consumption volume	1.436 l/h 87.6 cu.in./hr.	2.084 l/h 127.9 cu.in./hr.
Fuel consumption by weight	1.240 kg/h 2.73 lb./hr.	1.800 kg/h 3.97 lb./hr.
Specific fuel consumption	0.275 kg/HP-h .6 lb./HP.-hr.	0.3 kg/HP-h .65 lb./HP.-hr.
Effective horsepower	4.5 HP 4.4 HP.	6.0 HP 5.9 HP.
Mean effective piston pressure	3.75 atm.	5.0 atm.
Thermal efficiency for 10,000 calories per kg (1143 B.t.u./lb.)	0.23	0.21

The combustion was perfect and the fuel consumption satisfactory for an air-cooled engine of low power.

Fig. 57 shows a Frey and Fischer small engine with mechanical injection mounted on a motorcycle. It develops 6 HP. at n = 2400, with 90 mm (3.54 in.) stroke and 80 mm (3.15 in.)bore, and is said to have a fuel consumption of 250 g (0.55 lb.) per HP. ("Motorwagen" 1924, p.28).

No further details of this engine, of the Dorner engine, nor of the Aero engine have yet been published. It doubtless follows, however, from the information thus far published, that the right way has been adopted for greatly extending the use of the Diesel engine.

If, in the present article, the discussion of the Diesel engine, its manner of functioning and its most important parts have occupied much space, the reason is that, according to all indications, it is the particular oil engine which is not only able to use native fuels with great economy, but is, above all, best adapted to the severe operating conditions of vehicular traffic. The history of its development is brief (dating really from the time when the inadequacies of heavy-oil combustion by the explosion method were recognized) and is by no means closed. Important physical and chemical phenomena are yet little understood and fundamental questions are still strongly disputed. That difficulties have nevertheless been overcome, which seemed insurmountable six or seven years ago, is an accomplishment that cannot be overrated. Although at that time noted specialists considered the compressorless engine only conditionally utilizable, even as a stationary engine, there is today hardly any doubt that the production of a light Diesel engine suitable for aircraft lies within the realm of the technically attainable. According to American sources of information ("Mechanical Engineering," 1925, p.789, Heavy Oil Engines), the Eastern Engineering Corporation, Ltd., of Montreal, has produced from the designs of A. C. Attendu, a Diesel engine for aircraft, concerning which more details were given at a recent meeting of the Society of Automotive Engineers. It is a two-cylinder, two-stroke-cycle engine with a 140 mm (5.51 in.) bore and a 160 mm (6.3 in.) stroke. It has a step

piston, which delivers the scavenge air to the cylinders through mechanically operated valves. The exhaust is regulated by valves and ports. The engine weighs about 175 kg (386 lb.) and works with mechanical atomization at relatively high injection pressures, said to reach 420 atm. (about 6000 lb./sq.in.). After the engine had run 150 hours without a misfire in the Aeronautical Engine Laboratory of the Naval Aircraft Factory at Philadelphia, it was accepted by the U. S. Navy Department in November, 1925. The engine did not, however, develop 100 HP. at n = 1800 R.P.M., corresponding to 1.75 kg (3.86 lb.) per HP., as mentioned in the contract, its maximum output being 85 B.HP. at n = 1620, corresponding to 2.06 kg (4.54 lb.) per HP. By increasing the revolution speed to over 2000 R.P.M., it is hoped to increase the output to 125 HP. and to lower the power loading correspondingly. This result will be of far-reaching importance for aviation if further experiments, to be made on airplanes, demonstrate the perfect availability of the Diesel engine for this purpose. The small fuel consumption of the Diesel engine would increase the radius of action of the airplane, while the lower fuel cost would increase its economy. Above all, however, the use of a fuel which is not inflammable at ordinary temperatures, instead of gasoline, would mean a considerable reduction in the fire hazard and an almost complete elimination of fires which so often result in catastrophies, especially on airships.

What form the future development of the light high-speed

engine will take, cannot yet be predicted. I am therefore simply expressing my personal opinion when I predict that the two-stroke-cycle will determine the future of the small Diesel engine for vehicles. Here it offers considerably fewer difficulties than in large engines, where the high temperatures, which increase rapidly with the revolution speeds, greatly impair the lubrication and the durability of the materials. The way is here indicated, however, for offsetting to some extent the weight increase of the Diesel engine, necessitated by the higher working pressures, by doubling the effective working strokes, so that at most, it will not be heavier than a carburetor engine. While in the use of the two-stroke-cycle on the latter (in spite of preliminary air injection and other preventive measures, a mixing of the new charge with the residual exhaust gases and a scavenging with fresh gas cannot be avoided, the scavenging in the Diesel engine is done with air, thereby avoiding any waste of fuel. For less exacting uses and cheaper engines without very high revolution speeds, a two-cylinder engine with crank case scavenge pump will suffice. The low volumetric efficiency of this scavenge pump must, however, be improved, for which purpose (since step pistons are excluded on account of the impossibility of making them air-tight) displacer pistons are the best suited. These pistons improve the effective overpressure of the crank case, by means of special connecting rods attached to the crank shaft or linked to the regular connecting rods.

For higher revolution speeds, it will be sought to avoid the disadvantages of the crank case scavenge pump (subdivision of crank case, careful air-tightening of crank shaft by long journal bearings, etc.), by using a special scavenge pump, preferably a centrifugal or rotary pump. In comparison with the use of a crank case scavenge pump, this arrangement shortens the engine and enables the use of roller bearings for the crank shaft and connecting rods. If we choose a three-cylinder arrangement, we obtain a well-balanced engine free from inertia forces and tilting moments, which functions smoothly and satisfactorily without any valve gear. The arrangement of the U-shaped combustion space, which has given excellent results both in small air-cooled two-stroke carburetor engines and in large ship engines, must also be taken into consideration, all the more because the double-piston crank shaft drive affords very favorable kinetic conditions for the operation of the inlet and outlet ports. In choosing the scavenge pressure, one must not be too saving. It is better to use more power for supplying the scavenge air, provided the excess pressure can be used for producing the desired vortical motion. By the right choice of the scavenge pressure and the suitable construction of the inlet ports, the air can be given the desired direction and velocity from the start. This velocity can then be utilized, in conjunction with the kinetic energy of the fuel jet, for effecting a rapid combustion.

Whether and to what extent overloading can be used advantag-

eously for the purpose of permanent or temporary improvement and whether the energy of the exhaust gases is further utilizable in exhaust-gas turbines, are questions which have not been sufficiently solved to be answered here.

Many of the questions raised here are problems which have long engaged the attention of automobile and aircraft engine designers. Experimentation has been carried on in almost all of these fields. No systematic compilation and economic evaluation of all this work has, however, yet been made. Experimental results of great scientific and practical value slumber in the reports of the various researches, without (for economical or other reasons) being able to participate in progressive development. At the very time when the cooperation of all investigators is extremely urgent, there is a painful aloofness and commercial secretiveness. The development of the Diesel automotive engine might have made greater progress had not so many automobile firms been indifferent or skeptical regarding this new field. This skepticism is probably a heritage from the time of the heavy-oil carburetor engines, whose unsatisfactory performances brought numerous complaints from their users. As regards the modern high-speed Diesel engine, this skepticism is out of place. Naturally the development of this engine is a task which can be accomplished only by comprehensive and persistent work in the laboratory and on the test stand, but the attainable results fully justify all these efforts. So let it only be remembered

that in view of the great interest taken in the use of crude petroleum for automotive engines not only in Germany but also in Eastern Europe and in the United States, the possibility of exporting commercial vehicles equipped with Diesel engines might result in a great revival of our export trade. Now that out of the many conceivable solutions, a few practical engines have emerged, it becomes the task of the automobile industry to develop them further. It is in a better position to do this, because it has always owed its success less to large-scale researches than to its thorough absorption in structural details. Moreover, in its present stage, the development of the Diesel automotive engine cannot dispense with such intelligent cooperation. Now that the constructor of large stationary Diesel engines has evolved the small automotive engine, it is the task of the automobile manufacturer to develop a reliable practical engine for quantity production. The inevitable objection that automobile manufacturers now have neither time nor means for experimental research cannot be regarded as valid. Certainly a reasonable improvement in our production methods is a problem of far-reaching importance. Of no less importance, however, is our moral duty not only to preserve the legacy of an Otto, a Diesel, or a Maybach, but also to develop and improve this legacy. Activity in experiment and research is like a bill of exchange, the discounting of which lies in the future. If we refrain from this risk, we voluntarily exclude ourselves from association with those who

control the markets of the world. If we succeed, however, in uniting the intellectual forces of Germany in a common cause, we may expect to regain our former prestige in the world market.

Translation by Dwight M. Miner,
National Advisory Committee
for Aeronautics.

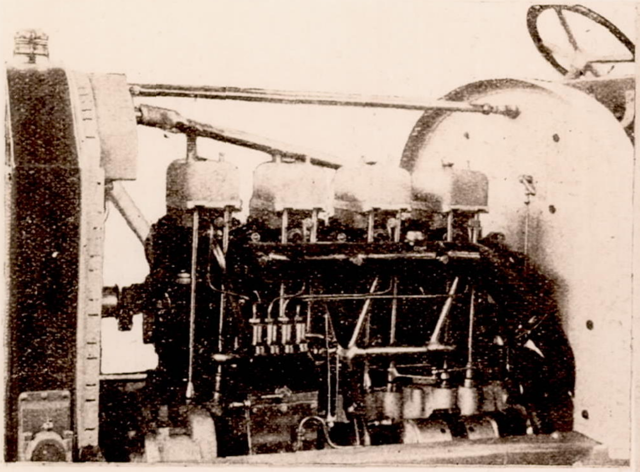

Fig.27 Benz heavy-oil engine for vehicles.

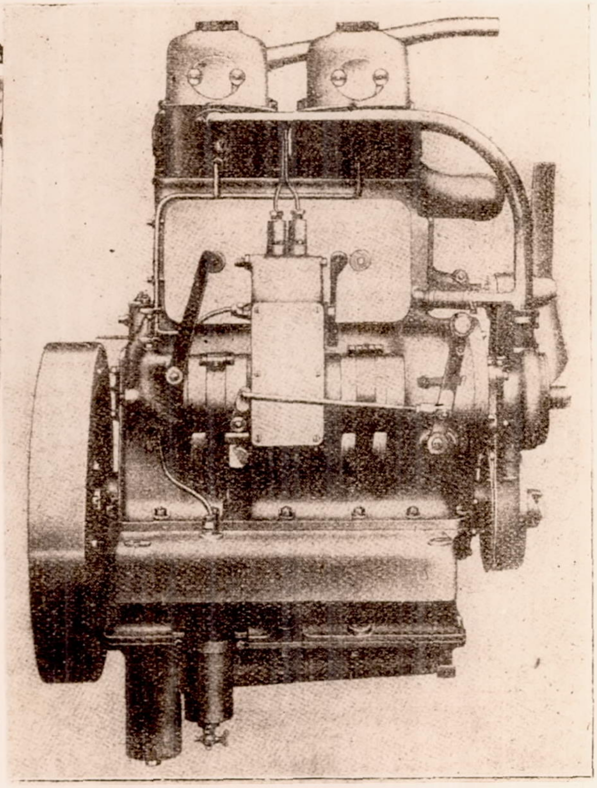

Fig.28 Benz-Sendling heavy-oil engine (Fuel-pump side).

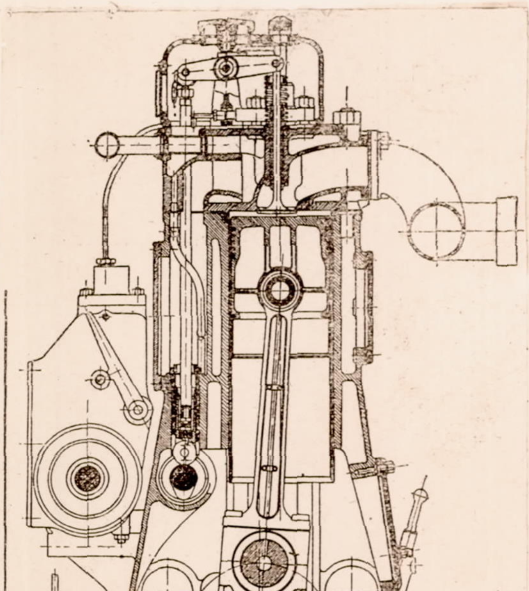

Fig.29 Section of Benz-Sendling heavy-oil engine

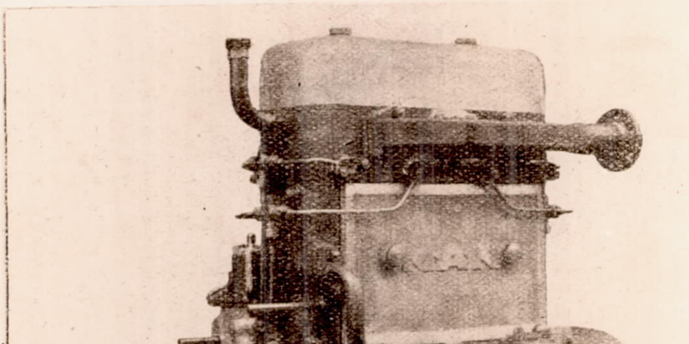

Fig.30 M.A.N. heavy-oil engine for vehicles.

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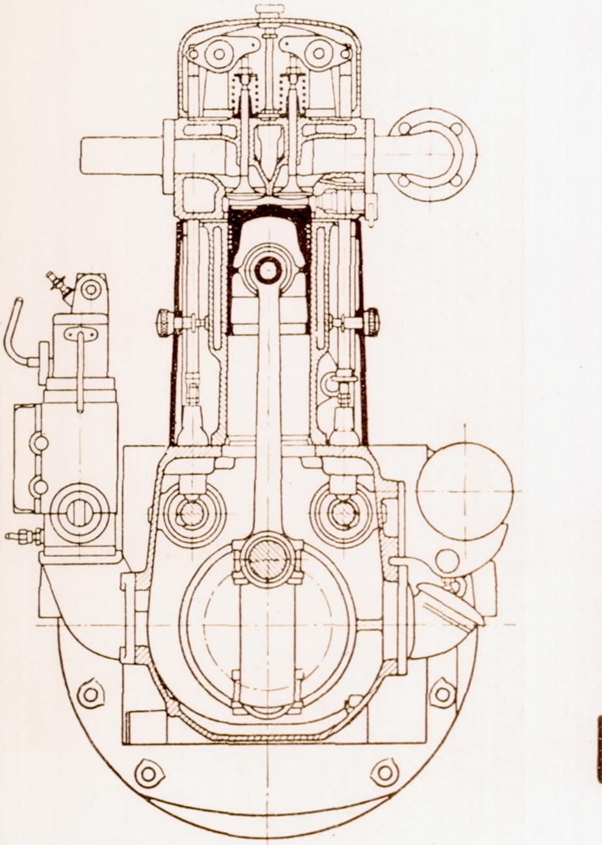


Fig.31 Section of M.A.N. heavy-oil engine for vehicles.

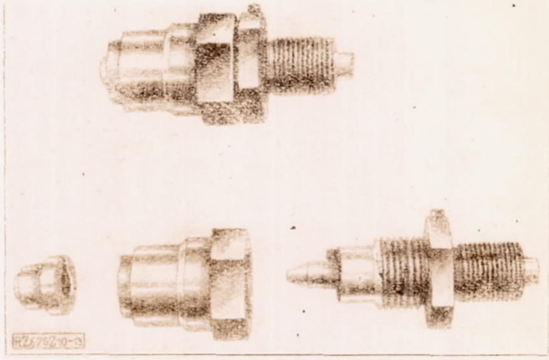


Fig.32 Nozzle of M.A.N. heavy-oil engine.

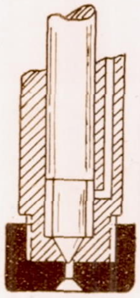


Fig.33 Closed one-hole nozzle

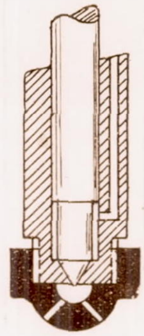


Fig.34 Closed multi-hole nozzle

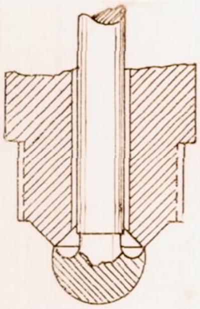


Fig.35 McKechnie nozzle

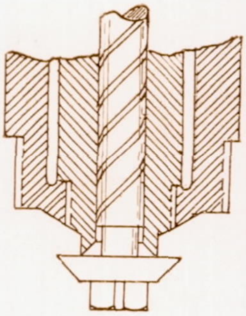


Fig.36 Peugeot-Tartrais whirling-spray nozzle

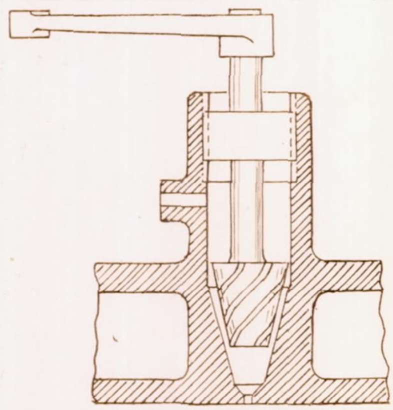


Fig.37 Nozzle of Lanz engine "Bulldog"

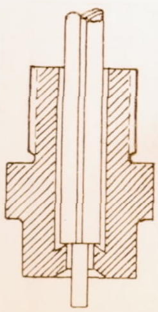


Fig.38 Petersen annular nozzle

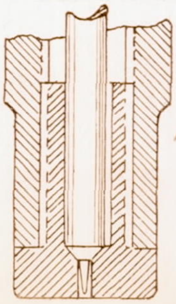


Fig.39 Benz annular nozzle

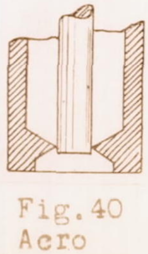
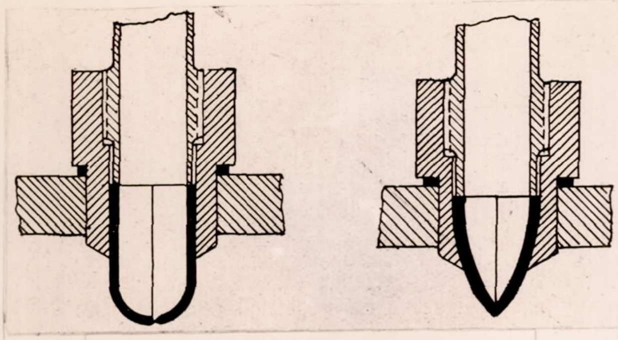


Fig.40 Acro nozzle



Figs.41 & 42 Hannover "Waggonfabrik" nozzle

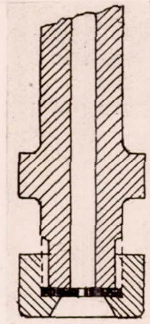


Fig.43 Open one-hole nozzle

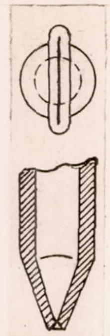


Fig.44 "Acro" nozzle

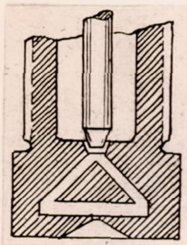


Fig.45 Brenkert nozzle

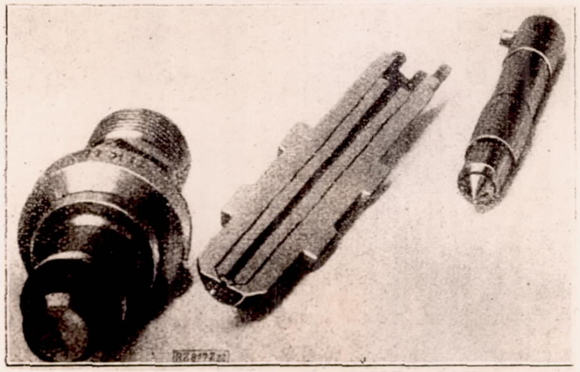


Fig.46 Nozzle of Junkers counter-piston engine

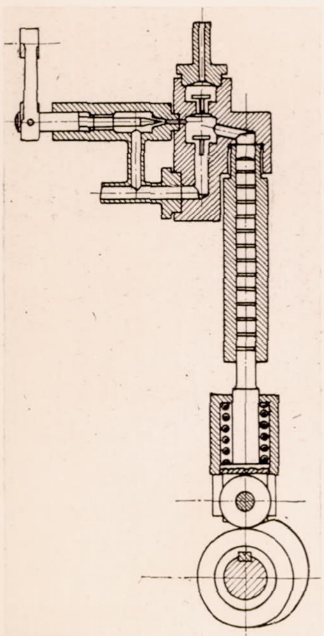


Fig.47 Fuel pump with throttle regulation.

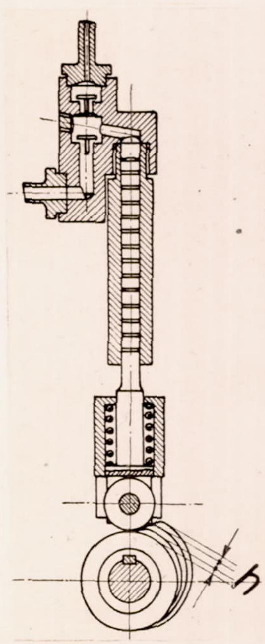


Fig.48 Fuel pump with variable stroke (oblique cam)

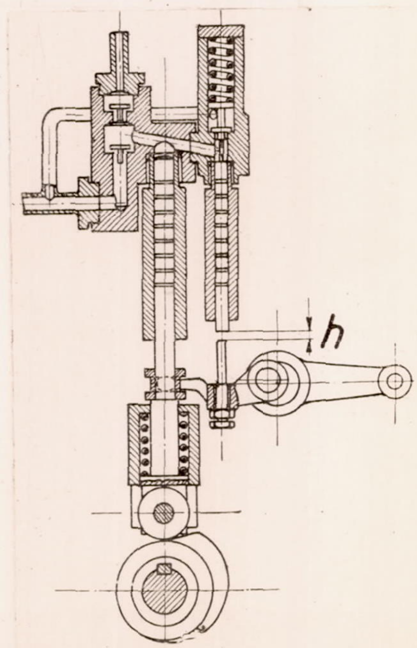


Fig.49 Fuel pump with overflow valve.

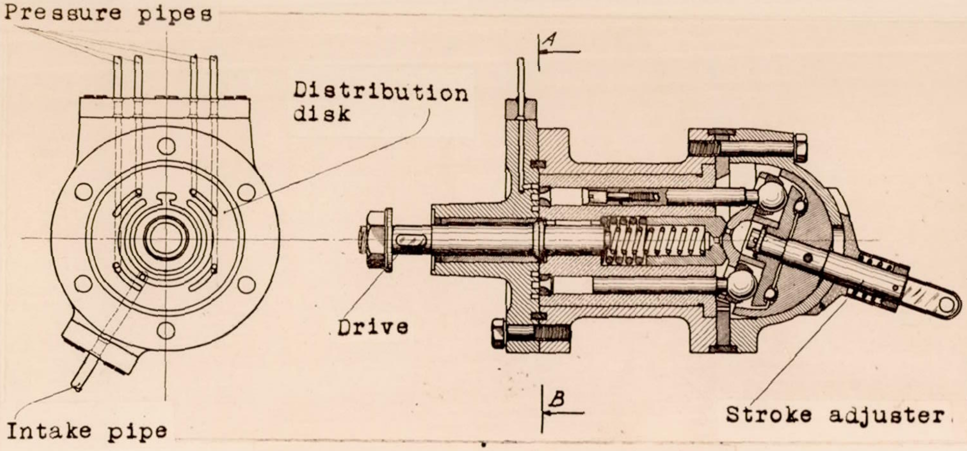


Fig.50 Egersdörfer fuel pump.

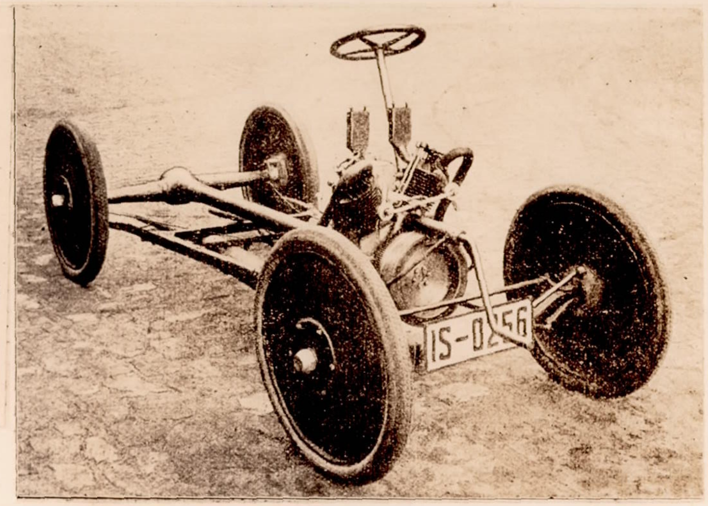


Fig.56 Motor vehicle with Dorner heavy-oil engine.

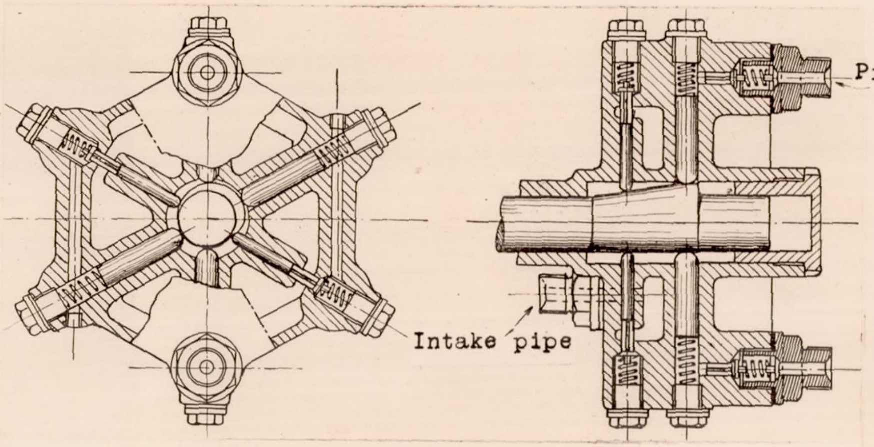


Fig.51 Rumpler fuel pump.

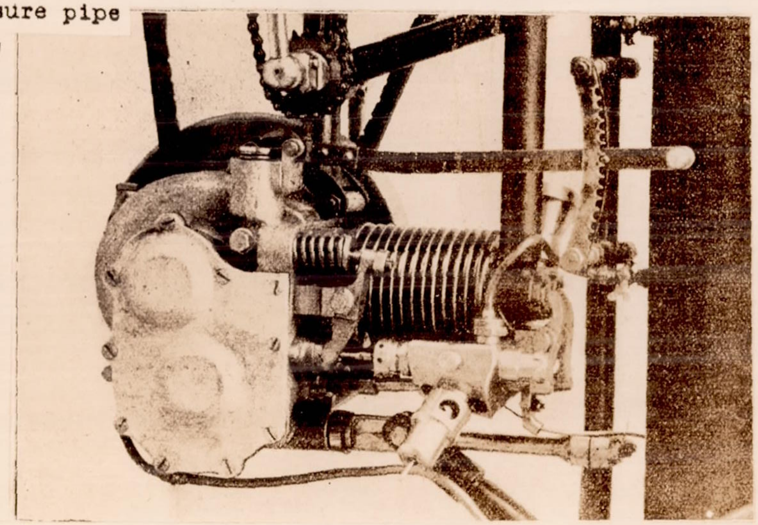


Fig.57 Frey-Fischer oil engine for motorcycles.

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a, at n = 1200 c, at n = 950

b, at n = 1050 d, at n = 800

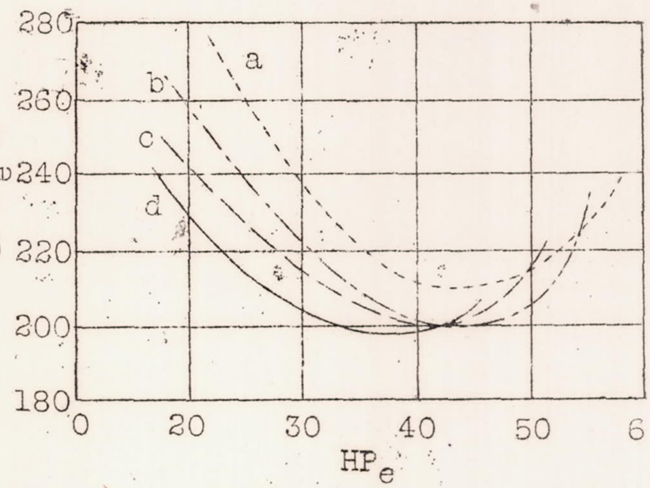

Fig. 52 Fuel consumption of M. vehicle engine.

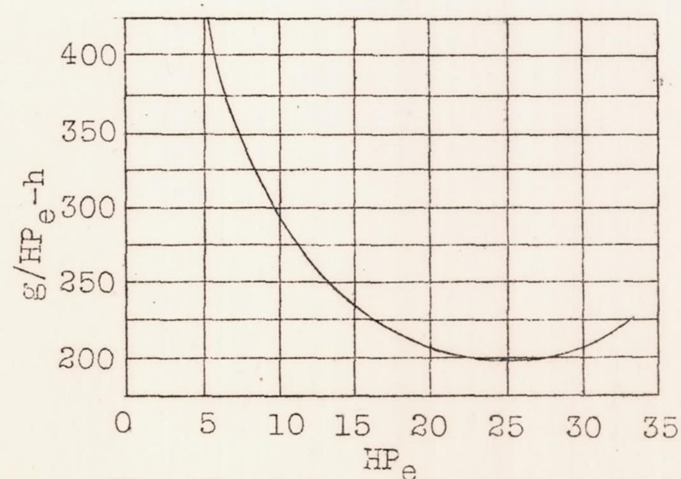

Fig.53 Fuel consumption of Benz-Sendling engine.

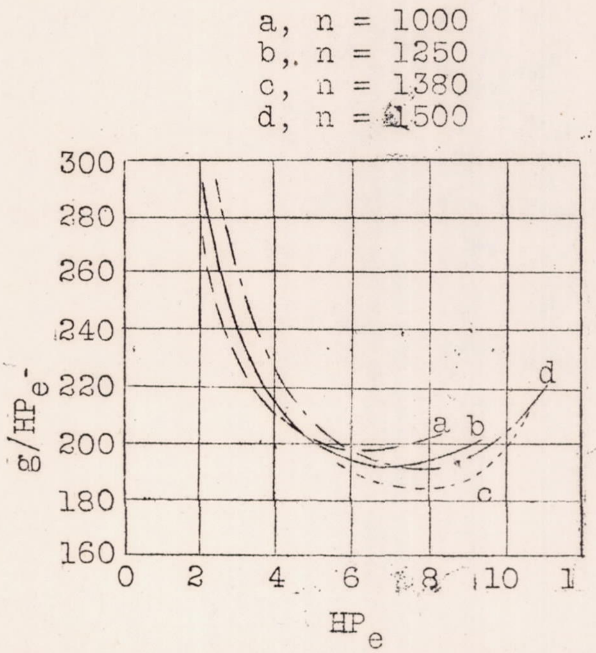

Fig.54 Fuel consumption of Acro experimental engine.

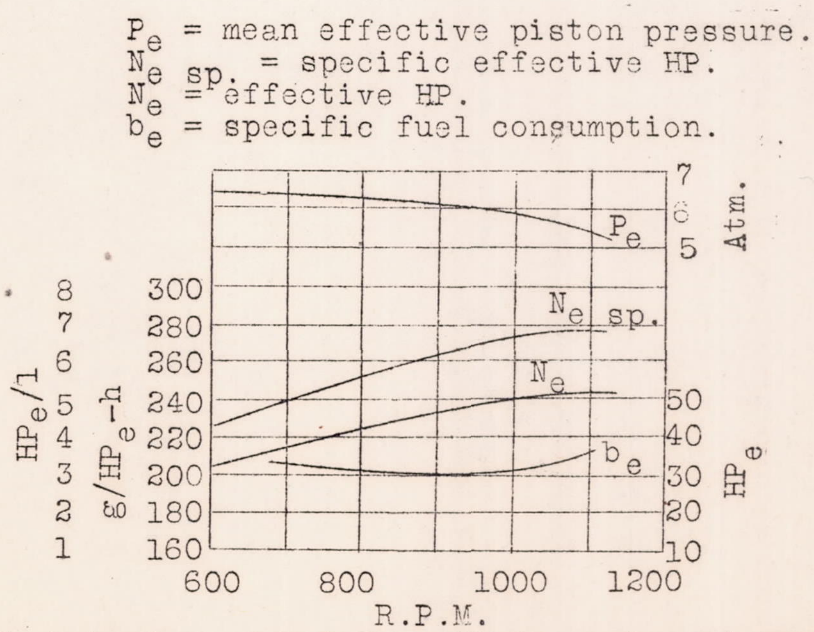

Fig.55 Behavior of M.A.N. vehicle engine at various revolution speeds and maximum torque.